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CERTIFICATION

I, Takao Kohno; 4-3, Tsuriganecho 2-chome, Chuo-ku, Osaka 540, Japan, hereby certify that I am the translator of the documents in respect of an application for a patent filed in Japan on the 30th day of November, 1998

and certify that the following is a true and correct translation to the best of my knowledge and belief.

KOHNO PATENT OFFICE

KOHNO Takao

Dated this 1st day of April, 2003

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[Name of the Document] Specification

[Title of the Invention] Hydraulic Control Valve and Power Steering Apparatus Using the Same

[Claims of Patent]

[Claim 1] A hydraulic control valve comprising a valve body having a plurality of first oil grooves disposed at equal distances from one another, extending in a longitudinal direction in an inner circumference of the valve body, in which: a valve spool is fitted allowing relative angular displacement, said valve spool having a plurality of second oil grooves provided on an outer peripheral surface of said valve spool alternately with respect to said first oil grooves; throttle portions are provided so that each of the throttle portions is provided in each gap between edge portions, at both sides in widthwise direction, of each of the first and second oil grooves which are adjacent to each other in a peripheral direction of the valve spool; and ones of said first and second oil grooves alternately act as oil supply chambers and oil discharge chambers, whereas the others act as oil feed chambers between said oil supply chambers and oil discharge chambers; characterized by comprising chamfer portions for adjusting throttle areas, provided on ones of the edge portions of said first and second oil grooves which form throttle portions between said oil supply chambers and oil feed chambers or which form said throttle portions between said oil discharge chamber and oil feed chambers.

[Claim 2] The hydraulic control valve according to claim 1, wherein said chamfer portions are provided in the throttle portions where a flow is generated from each of said second oil grooves toward each of said first oil grooves.

[Claim 3] A power steering apparatus, comprising a hydraulic pump, being driven by an electric motor for supplying oil pressure to a hydraulic cylinder for steering assistance; and a hydraulic control valve for switching supply of pressurized oil from said hydraulic pump to said hydraulic cylinder, characterized by comprising the hydraulic control valve described in claim1 or claim 2.

[Claim 4] A power steering apparatus, comprising: a hydraulic pump, being driven by an electric motor for supplying oil pressure to a hydraulic cylinder for steering assistance; and hydraulic control valve comprising a valve body having a plurality of first oil grooves disposed at equal distances from one another, extending in a longitudinal direction in an inner circumference of the valve body, in which: a valve spool is fitted allowing relative angular displacement, said valve spool having a plurality of second oil grooves provided on an outer peripheral surface of said valve spool alternately with respect to said first oil grooves; throttle portions are provided so that each of the throttle portions is provided in each gap between edge portions, at both sides in widthwise direction, of each of the first and second oil grooves which are adjacent to each other in a peripheral direction of the valve spool; and ones of said first and second oil grooves alternately act as oil supply chambers and oil discharge chambers, whereas the others act as oil feed chambers between said oil supply chambers and oil discharge chambers; and said hydraulic control valve switches supply of operating oil from said hydraulic pump to said hydraulic cylinder, characterized in that said hydraulic control valve comprises chamfer portions for adjusting throttle areas, provided on ones of the edge portions of said first and second oil grooves which form throttle

portions between said oil supply chambers and oil feed chambers or which form said throttle portions between said oil discharge chamber and oil feed chambers, and said hydraulic pump is driven such that a flow rate becomes low or zero at a time of standby control, and such that the flow rate becomes high in accordance with steering angular velocity when steering is carried out.

[Detailed Description of the Invention]

[0001]

[Field of Industrial Application]

The present invention relates to a rotary-type hydraulic control valve for controlling an oil pressure utilizing relative angular displacement between a valve body and a valve spool which are coaxially fitted to each other for relative rotation, and the present invention also relates to a hydraulic-type power steering apparatus for controlling the oil pressure to be supplied to a steering-assist hydraulic cylinder in accordance with the steering operation using the hydraulic control valve.

[0002]

[Prior Art]

In a hydraulic power steering apparatus which assists the steering operation by hydraulic force generated by a plurality of hydraulic cylinders (power cylinders) disposed in the steering apparatus for reducing labor required for the operation of a steering wheel, a hydraulic control valve for controlling supply and discharge of oil pressure in accordance with direction and magnitude of a steering torque applied to the steering wheel is disposed between, a hydraulic pump driven by an electric motor and a hydraulic tank for

accommodating operating oil and the hydraulic cylinders, and pressurized oil generated by the hydraulic pump is supplied to two cylinder chambers of the hydraulic cylinders by the operation of the hydraulic control valve.

[0003]

As for the hydraulic control valve, a rotary-type hydraulic control valve directly utilizing the rotation of the steering wheel is used. In this rotary hydraulic control valve, an input shaft connected to the steering wheel and an output shaft connected to the steering mechanism are coaxially connected to each other through a torsion bar, a valve spool integrally formed with one of the connected ends is coaxially and relatively rotatably fitted into a cylindrical valve body engaged with the other connected end, and when the steering torque is applied to the steering wheel, the torsion bar is twisted and relative angular displacement is generated between the valve body and the valve spool.

[0004]

FIG. 7 and FIG. 9 are schematic transverse sectional views showing examples of the structure of a hydraulic control valve described in Japanese Patent Application Laid-open No. H9-39814(1997).

A plurality of first oil grooves 4 extending in the longitudinal direction are disposed along the circumferential direction in an inner peripheral surface of a valve body 1 at even intervals. A plurality of second oil grooves 5 are alternatively disposed in an outer peripheral surface of a valve spool 2 with respect to the first oil grooves 4. A plurality of throttle portions 6a, 6b are provided in gaps between edge

portions, at both sides in widthwise direction, of the first and second oil grooves 4 and 5 adjacent to each other in a peripheral direction of the valve spool, and ones of said first and second oil grooves alternately act as oil supply chambers 10 and oil discharge chambers 11, whereas the others act as oil feed chambers 12 and 13 between said oil supply chambers 10 and oil discharge chambers 11. The oil supply chambers 10 are in communication with discharge side of the hydraulic pump P as an oil pressure source and oil discharge chambers 11 are in communication with an oil tank T which receives the oil. The oil feed chambers 12 and 13 are alternately in communication with cylinder rooms SR and SL of a hydraulic cylinder S which receives the oil.

[0005]

In the example shown in FIG. 7, the first oil grooves 4, 4, ... parallelly arranged in the inner peripheral surface of the valve body 1 form first oil feed chambers 12, 12, ... and second oil feed chambers 13, 13. The first oil feed chambers 12, 12, ... are in communication of a right side cylinder chamber SR of a hydraulic cylinder S to which oil is fed through an oil feed hole formed in the valve body 1. The second oil feed chambers 13, 13, ... are in communication with a left side cylinder chamber SL of the hydraulic cylinder S through an oil feed hole formed in the valve body 1. Further, the second oil grooves 5, 5, ... parallelly arranged in the outer peripheral surface of the valve spool 2 alternately form oil supply chambers 10, 10, ... which are in communication with discharge side of a hydraulic pump P as an oil pressure source through an oil introducing hole formed in the valve body 1 and oil discharge chambers 11, 11, ... which are in communication with an oil tank T to receive the oil through an oil

discharge hole formed in the valve spool 2.

[0006]

In an example shown in FIG. 9, second oil grooves 5, 5, ... parallelly arranged in an outer peripheral surface of a valve spool 2 alternately form first oil feed chambers 12, 12, ... and second oil feed chambers 13, 13, ... The first oil feed chambers 12, 12, ... are in communication a right side cylinder chamber SR of a hydraulic cylinder S to which oil is fed through an oil feed hole formed in the valve body 1. The second oil feed chambers 13, 13, ... are in communication with a left side cylinder chamber SL of the hydraulic cylinder S through an oil feed hole formed in the valve body 1. First oil grooves 4, 4, ... parallelly arranged in an inner peripheral surface of the valve body 1 alternately form oil supply chambers 10, 10, ... which are in communication with discharge side of a hydraulic pump P through an oil introducing hole formed in the valve body 1 and oil discharge chambers 11, 11, ... which are in communication with an oil tank T through an oil discharge hole formed in the valve spool 2.

[0007]

The first oil grooves 4, 4, ... and the second oil grooves 5, 5, ... are in communication to one another while having equal areas in gaps between edge portions, at both sides in widthwise direction, of the first and second oil grooves, and the communicating portions act as throttle portions 6a, 6b whose throttle areas are changed in accordance with the relative angular displacement. Therefore, oil pressure to be supplied to the cylinder chambers SR, SL through the first and second oil feed chambers 12, 13 are controlled by the change in throttle areas of these throttle portions 6a, 6b.

[0008]

FIGS. 8 (a) and 8(b) are explanatory views of operation of an oil feed chamber, an oil supply chamber, and an oil discharge chamber shown in straightly developed manner, which are arranged along a fitted peripheral surfaces of a valve body and a valve spool in the conventional hydraulic control valve. FIG. 8(a) shows a state in which the relative angular displacement is not generated between the valve body 1 and the valve spool 2. In this state, the pressurized oil supplied from the hydraulic pump P to the oil supply chambers 10 flows such that the oil is equally divided into the first and second oil feed chamber 12, 13 which are adjacent to each other through the throttle portions 6a, 6a having the equal sectional areas in the circumferential direction on the both sides of the oil supply chambers 10 and are introduced into the oil discharge chambers 11, 11 through the throttle portions 6b, 6b having the equal sectional areas in the circumferential direction on the both sides of the first and second oil feed chamber 12, 13, and flows into the oil tank T which is in communication with these oil discharge chambers 11, 11. Therefore, the oil pressure supplied to the oil supply chambers 10 is not fed to any of the cylinder chambers SR, SL, and no force is generated in the hydraulic cylinder S.

[0009]

FIG. 8(b) shows a state in which the steering torque is applied to the steering wheel, and the relative angular displacement is generated between the valve body 1 and the valve spool 2. In this state, the throttle area of one of the throttle portions 6a, 6a on the both sides of the oil supply chamber 10 (the first oil feed chamber 12 side) is

increased, and the throttle area of the other throttle portion 6a (the second oil feed chamber 13 side) is reduced. As a result, the pressurized oil supplied to the oil supply chamber 10 is introduced mainly to the first oil feed chamber 12 through the throttle portion 6a whose throttle area is increased. That is, a pressure difference is generated between the first oil feed chamber 12 and the second oil feed chamber 13, i.e., between the cylinder chambers SR, SL which are in communication with the first oil feed chamber 12 and the second oil feed chamber 13, and the hydraulic cylinder S generates hydraulic force (steering assist force) corresponding to this pressure difference.

[0010]

The pressure difference generated at that time depends on a degree of reduction in the throttle area of the other throttle portion 6a (the oil feed chamber 13 side), and the degree of the reduction corresponds to magnitude of the relative angular displacement, i.e., the magnitude of the steering torque applied to the steering wheel. Therefore, the force generated by the hydraulic cylinder S has a direction and magnitude corresponding to the steering torque, and can assist the steering operation. At that time, oil in the left cylinder chamber SL pushed out by the operation of the hydraulic cylinder S is circulated into the second oil feed chamber 13, and is introduced into the oil discharge chamber 11 through the throttle portion 6b whose throttle area is increased on the one side of the second oil feed chamber 13, and is discharged into the oil tank T which is in communication with the oil discharge chamber 11.

[0011]

Preferable increasing characteristics of the steering assist force

in the power steering apparatus are not characteristics to increase in proportion to the steering torque, but are characteristics in which the steering assist force is gradually increased in a range having small steering torque but the force is rapidly increased when the steering torque exceeds a predetermined limit. To obtain such characteristics, chamfer portions 7, 7, ... each having a predetermined inclination angle with respect to a peripheral surface of the valve spool 2 and having a predetermined width in the circumferential direction are provided on the edge portions of all of the second oil grooves 5, 5, ... of the valve spool 2. With these chamfer portions, the throttle areas of the throttle portions 6a, 6b are slowly changed with respect to the relative angular displacement between the valve body 1 and the valve spool 2.

[0012]

In the hydraulic control valve operating as described above, four, six, eight or more first and second oil grooves 4, 5 are equally disposed, half of one of the first and second oil grooves 4 and 5 are formed as the oil supply chambers 10, and the remaining half are formed as the oil discharge chambers 11.

[0013]

In a hydraulic control valve in which four oil grooves are disposed at equal distances from one another, since the oil supply chambers and oil discharge chambers are provided two each, the flow rate of the pressurized oil introduced into one oil supply chamber 10 can be relatively large. On the other hand, since the two oil supply chambers 10 whose oil pressure become high because they control the hydraulic cylinder S are disposed with phase difference of 180 degrees,

balance of pressure distribution applied to the valve body 1 is inferior, and the valve body 1 is deformed into elliptic shape. At that time, a fitting gap of about 10μm between the valve body 1 and the valve spool 2 is changed, and there is an adverse possibility that biting phenomenon is generated between the valve body 1 and the valve spool 2.

[0014]

In a hydraulic control valve in which six oil grooves are disposed at equal distances from one another, the oil supply chamber 10 and oil discharge chamber 11 are provided three each, and in a hydraulic control valve in which eight oil grooves are disposed at equal distances from one another, the oil supply chamber 10 and oil discharge chamber 11 are provided four each. Therefore, the flow rate of the pressurized oil distributed to one oil supply chamber 10 is smaller than that of the hydraulic control valve in which four oil grooves are disposed at equal distances. However, in the hydraulic control valve in which six oil grooves are disposed at equal distances, the oil supply chambers 10 whose oil pressure become high are disposed at three locations with phase difference of 120 degrees, and in the hydraulic control valve in which eight oil grooves are disposed at equal distances, the oil supply chambers 10 whose oil pressure become high are disposed at four locations with phase difference of 90 degrees. Therefore, the balance of the pressure distribution applied to the valve body 1 is excellent, the deformation of the valve body 1 is restrained, and the fitting gap between the valve body 1 and the valve spool 2 is excellently held. That is, it is preferable to dispose at least six oil grooves in the hydraulic control valve.

[0015]

The present applicant is developing a power steering apparatus in which the conventional power steering apparatus with the above-described structure is used, a hydraulic pump is standby controlled (low-speed rotation or zero-speed rotation), and when a steering torque is not applied to a steering wheel at the time of idling or the like, the small or zero flow rate of pressurized oil about 1 to 2 liter/min is introduced into the oil supply chamber of the hydraulic control valve, the steering angle of the steering wheel is detected, and the flow rate of oil of the hydraulic pump can be transitionally increased in accordance with the steering angular velocity based on the detected steering angle. According to such a power steering apparatus, it is possible to abruptly change the flow rate to be controlled of the hydraulic control valve from the small flow rate as small as possible or zero flow rate to high flow rate as compared with the conventional small flow rate.

[0016]

FIG. 10 is a view showing the flow rate characteristics showing a relation between steering angular velocity and the flow rate of a pump, and FIG. 11 is a view showing hydraulic characteristics of the conventional hydraulic control valve showing a relation between input torque applied to the steering wheel and hydraulic force controlled by the hydraulic control valve.

In FIG. 10, the number of revolution of an electric motor for the hydraulic pump is transitionally increased as the steering angular velocity is increased, and the flow rate of the hydraulic pump is transitionally increased. In FIG. 11, the increase in the input torque

is reduced as the hydraulic force controlled by the hydraulic control valve is increased.

[0017]

[Problems to Be Solved by the Invention]

However, the conventional hydraulic control valve constructed as above is designed or produced such that the flow rate (minimum flow rate to be controlled) at the time of standby control becomes the small flow rate of about 1 to 2 liter/min or zero flow rate as described above. Therefore, when the conventional hydraulic control valve is applied to the hydraulic control valve under development, the oil pressure characteristics in a region where the flow rate to be controlled is minimum which is smaller than the conventional small flow rate become unstable. That is, when the oil pressure rises transitionally at the beginning of steering operation, there are problems that the hydraulic characteristics becomes abruptly discontinuous, and the steering torque becomes discontinuous.

[0018]

FIG. 12 is a view for explaining a state where the hydraulic characteristics of the conventional hydraulic control valve become discontinuous.

Here, it is supposed that the conventional hydraulic control valve whose hydraulic characteristics in the small flow rate region is varied largely and which has six or eight oil grooves disposed at equal distances from one another is applied to the hydraulic control valve under development as it is. That is, in the case of the hydraulic control valve having eight oil grooves disposed at equal distances from one another, the flow rate to be distributed to each of the throttle

portions is reduced as small as 0.125 liter/min or less, and this small flow rate causes large variation in the hydraulic characteristics, and this is controlled by the chamfer portion where the hydraulic characteristics are unstable and therefore, the rise of the oil pressure characteristics at the beginning of the steering operation becomes unstable. Therefore, when the steering operation is started, jump is generated in the oil pressure characteristics as shown in FIG. 12, the hydraulic characteristics become discontinuous, and the steering torque becomes discontinuous.

[0019]

On the other hand, in the case where the hydraulic control valve having four oil grooves disposed at equal distances from one another is applied to the hydraulic control valve under development as it is, the flow rate to be distributed to each of the throttle portions is increased as compared to a case in which a hydraulic control valve having six or more oil grooves disposed at equal distances from one another, but the balance of pressure distribution applied to the valve body is inferior, the biting phenomenon is generated between the valve body and the valve spool and thus, it is not preferable to apply the hydraulic control valve having the four oil grooves disposed at equal distances from one another.

[0020]

The present invention has been accomplished to solve the above-described problems, and it is one object of the invention to provide a hydraulic control valve in which the chamfer portions are provided only in the throttle portions between the oil supply chambers and the oil feed chambers or the chamfer portions are provided only in

the throttle portions between the oil discharge chamber and the oil feed chambers so as to stabilize the hydraulic characteristics, when the minimum flow rate to be controlled is reduced as small as possible, and to eliminate the discontinuity of the hydraulic characteristics in a region where the flow rate to be controlled is minimum, even if an existing hydraulic control valve in which six or more oil grooves are disposed at equal distances from one another is adopted. It is another object of the invention to provide a hydraulic control valve having the chamfer portions in the throttle portions where flows are generated from the second oil grooves to the first oil grooves so as to reduce flowing noise caused by operating oil when a relative angular displace between the valve body and the valve spool is great. Further, it is another object of the present invention to provide a power steering apparatus including a hydraulic pump which is driven such that a flow rate becomes low or zero flow rate during the standby control, and such that the flow rate becomes high flow rate in accordance with steering angular velocity when steering operation is carried out. With this design, it is possible to reduce the energy consumption while the steering wheel is not operated at the time of idling for example without the discontinuity of the hydraulic characteristics.

[0021]

[Means for Solving the Problems]

A hydraulic control valve according to a first aspect of the present invention comprising a valve body having a plurality of first oil grooves disposed at equal distances from one another, extending in a longitudinal direction in an inner circumference of the valve body, in which: a valve spool is fitted allowing relative angular displacement,

said valve spool having a plurality of second oil grooves provided on an outer peripheral surface of said valve spool alternately with respect to said first oil grooves is; throttle portions are provided so that each of the throttle portions is provided in each gap between edge portions, at both sides in widthwise direction, of each of the first and second oil grooves which are adjacent to each other in a peripheral direction of the valve spool; and ones of said first and second oil grooves alternately act as oil supply chambers and oil discharge chambers, whereas the others act as oil feed chambers between said oil supply chambers and oil discharge chambers, is characterized by comprising chamfer portions for adjusting throttle areas, provided on ones of the edge portions of said first and second oil grooves which form throttle portions between said oil supply chambers and oil feed chambers or which form said throttle portions between said oil discharge chamber and oil feed chambers.

[0022]

In this aspect of the invention, chamfer portions are provided only on throttle portions between an oil supply chamber and oil feed chambers, or only throttle portions between an oil discharge chamber and oil feed chambers, and the number of portions where the flow rate is controlled by the chamfer portions is reduced to half as compared with the existing structure. As a result, even if six or more oil grooves of each of a valve body and a valve spool are disposed at equal distances from one another, when the minimum flow rate to be controlled is reduced as small as possible, a distribution amount for one portion where the flow rate is controlled can be doubled as compared with a conventional structure, it is possible to stabilize the

hydraulic characteristics in a region where the flow rate to be controlled is minimum, and to eliminate the discontinuity of the hydraulic characteristics. Accordingly, the flow rate of the hydraulic pump is increased on the basis of the steering angular velocity of the steering wheel so as to increase the flow rate of the hydraulic control valve from a flow rate as smaller as possible compared with the conventional minimum flow rate or zero flow rate to a high flow rate. With this structure, there is no discontinuity of the hydraulic characteristics in a region where the flow rate to be controlled is minimum, and it is possible to inexpensively produce a hydraulic control valve which eliminates discontinuity in the steering torque, by using an existing hydraulic control valve having six or more oil grooves disposed at equal distances from one another.

[0023]

A hydraulic control valve according to a second aspect of the invention is characterized in that said chamfer portions are provided in the throttle portions where a flow is generated from each of said second oil grooves toward each of said first oil grooves.

[0024]

According to the second aspect of the present invention, since it is possible to flow the operating oil introduced in the oil supply chamber while gradually and straightly reducing the pressure from the second oil groove toward the first oil groove along the chamfer, it is possible to reduce flowing noise which is generated when the relative angular displacement between the valve body and the valve spool is great. That is, as shown in FIG. 1, with a hydraulic control valve in which the first oil grooves of the valve body are formed as the oil feed

chambers and the second oil grooves are formed as the oil supply chambers and oil discharge chambers, flows from the second oil groove toward the first oil groove are generated in the throttle portions on the both sides of the oil supply chamber and flows from the first oil groove toward the second oil groove are generated in the throttle portions on the both sides of the oil discharge chamber. Moreover, as shown in FIG. 3, with a hydraulic control valve in which the second oil grooves of the valve spool are formed as the oil feed chambers and the first oil grooves of the valve body are formed as the oil supply chambers and oil discharge chambers, flows from the first oil groove toward the second oil groove are generated in the throttle portions on the both sides of the oil supply chamber, and flows from the second oil groove toward the first oil groove are generated in the throttle portions on the both sides of the oil discharge chamber. At that time, when high pressure oil flows from the second oil groove toward the first oil groove, the pressure is gradually and straightly reduced as the oil flows through the chamfer. Therefore, it is possible to eliminate the cavitation which causes flowing noise. On the other hand, in the case where a chamfer portion is formed in the throttle portions of on both sides of the oil discharge chamber of the hydraulic control valve shown in FIG. 1 and in the case where a chamfer portion is formed in the throttle portions of on both sides of the oil supply chamber of the hydraulic control valve shown in FIG. 3, as soon as the high pressure oil flows through the chamfer portion, the pressure is abruptly reduced generating the cavitation which causes flowing noise.

[0025]

A power steering apparatus according to the third aspect of the

invention comprising a hydraulic pump, which is driven by an electric motor for supplying oil pressure to a hydraulic cylinder for steering assistance, and a hydraulic control valve for switching supply of operating oil from said hydraulic pump to said hydraulic cylinder, is characterized by comprising the hydraulic control valve described in claim1 or claim 2.

[0026]

According to the third aspect of the invention, in the case where the hydraulic control valve is based on the first aspect of the invention, the flow rate of the hydraulic pump is increased on the basis of the steering angular velocity of the steering wheel so as to increase the flow rate of the hydraulic control valve from a flow rate as smaller as possible compared with the conventional minimum flow rate or zero flow rate to a high flow rate. With this structure, there is no discontinuity of the hydraulic characteristics in a region where the flow rate to be controlled is minimum, and it is possible to inexpensively produce a hydraulic control valve which eliminates discontinuity in the steering torque, by using an existing hydraulic control valve having six or more oil grooves disposed at equal distances from one another. Moreover, in the case where the hydraulic control valve is based on the second aspect of the invention, it is possible to reduce flowing noise which is generated when the relative angular displacement between the valve body and the valve spool is great.

[0027]

A power steering apparatus according to a fourth aspect of the invention is provided with: a hydraulic pump, being driven by an electric motor for supplying oil pressure to a hydraulic cylinder for

steering assistance; and hydraulic control valve comprising a valve body having a plurality of first oil grooves disposed at equal distances from one another, extending in a longitudinal direction in an inner circumference of the valve body, in which: a valve spool is fitted allowing relative angular displacement, said valve spool having a plurality of second oil grooves provided on an outer peripheral surface of said valve spool alternately with respect to said first oil grooves; throttle portions are provided so that each of the throttle portions is provided in each gap between edge portions, at both sides in widthwise direction, of each of the first and second oil grooves which are adjacent to each other in a peripheral direction of the valve spool; and ones of said first and second oil grooves alternately act as oil supply chambers and oil discharge chambers, whereas the others act as oil feed chambers between said oil supply chambers and oil discharge chambers; and said hydraulic control valve switches supply of operating oil from said hydraulic pump to said hydraulic cylinder, characterized in that said hydraulic control valve comprises chamfer portions for adjusting throttle areas, provided on ones of the edge portions of said first and second oil grooves which form throttle portions between said oil supply chambers and oil feed chambers or which form said throttle portions between said oil discharge chamber and oil feed chambers, and said hydraulic pump is driven such that a flow rate becomes low or zero at a time of standby control, and such that the flow rate becomes high in accordance with steering angular velocity when steering is carried out.

[0028]

According to the fourth aspect, the flow rate of the hydraulic

pump is increased on the basis of the steering angular velocity of the steering wheel so as to increase the flow rate of the hydraulic control valve from a flow rate as smaller as possible compared with the conventional minimum flow rate or zero flow rate to a high flow rate. With this structure, there is no discontinuity of the hydraulic characteristics in a region where the flow rate to be controlled is minimum, and it is possible to inexpensively produce a hydraulic control valve which eliminates discontinuity in the steering torque, by using an existing hydraulic control valve having six or more oil grooves disposed at equal distances from one another. Moreover, with this design, it is possible to reduce the energy consumption by stopping the operation of the electric motor while the steering wheel is not operated at the time of idling, for example.

[0029]

[Embodiments of the Invention]

The present invention will be explained below in detail based on the drawings showing embodiments of the invention. FIG. 1 is a schematic transverse sectional view showing a first embodiment of a hydraulic control valve according to the present invention, together with a hydraulic circuit of a power steering apparatus.

First Embodiment

In FIG. 1, numeral 1 represents a valve body, and numeral 2 represents a valve spool. Basic structures of these valve body 1 and valve spool 2 are the same as the conventional ones shown in FIG. 1. Eight first oil grooves 4, 4, ... having equal widths are parallelly disposed in an inner peripheral surface of the cylindrical valve body 1 at equal distances from one another in the circumferential direction.

Eight second oil grooves 5, 5, ... are parallelly disposed in an outer peripheral surface of the thick and cylindrical valve spool 2 having the outer diameter which is substantially equal to the inner diameter of the valve body 1 at equal distances in the circumferential direction.

[0030]

The valve spool 2 is relatively rotatably fitted into the valve body 1 coaxially, and the valve spool 2 and the valve body 1 are connected to each other by a torsion bar 3 which is inserted into the valve spool 2. The first oil grooves 4, 4, ... and the second oil grooves 5, 5, ... are placed alternatively in the circumferential direction as shown in the drawing in a neutral state in which the torsion bar 3 is not twisted, and the first oil grooves 4, 4, ... and the second oil grooves 5, 5, ... are positioned such that each groove is in communication with adjacent grooves at both sides.

[0031]

Each of the first oil grooves 4, 4, ... of the valve body 1 is opposed to a land between the second oil grooves 5, 5, ... of the valve spool 2, and each of the second oil grooves 5, 5, ... of the valve spool 2 is opposed to a land between the first oil grooves 4, 4, ... of the valve body 1. With such a structure, eight oil chambers inside (in a gaps between edge portions at both sides in widthwise direction) the first oil grooves 4, 4, ... and eight oil chambers outside (in a gaps between edge portions at both sides in widthwise direction) the second oil grooves 5, 5, ... are alternately arranged while having communication portions therebetween along the fitting peripheral surfaces between the valve body 1 and the valve spool 2.

[0032]

The relative angle between the valve body 1 and the valve spool 2 can be displaced within a range of twist of the torsion bar 3 which connects the valve body 1 and the valve spool 2 to each other, and the communication portion between each of the oil chambers, i.e., gaps between edge portions, at both sides in widthwise direction, of the first and second oil grooves 4 and 5 act as throttle portions 6a and 6b which increase or reduce a sectional area (throttle area) in the circumferential direction of each of the communication portion in accordance with the relative angular displacement.

[0033]

Among the eight oil chambers formed by the second oil grooves 5, 5, ... of the valve spool 2, alternate four of them pass through a peripheral wall of the valve body 1, and are brought into communication with a discharge side of a hydraulic pump P which is an oil pressure source through independent oil introducing holes having openings in outer sides of the second oil grooves 5, 5, ..., and form oil supply chambers 10, 10 ... to which pressurized oil is supplied from the hydraulic pump P. On the other hand, the remaining four oil chambers pass through the valve spool 2 in the radial direction, and are brought into communication with an oil tank T to which the oil is discharged through independent oil discharge holes having openings in bottoms of the second oil grooves 5, 5, ... and through a hollow portion in the valve spool 2, and form oil discharge chambers 11, 11, ... which act as passages for discharging oil into the oil tank T.

[0034]

On the other hand, among the eight oil chambers formed inside the first oil grooves 4, 4, ..., four of them which are adjacent to the

same side as the oil supply chambers 10, 10 ... in the circumferential direction pass through a peripheral wall of the valve body 1, are connected with a right side cylinder chamber SR of a hydraulic cylinder S to which the oil is fed through independent oil introducing holes having openings in bottoms of the first oil grooves 4, 4, ..., and form first oil feed chambers 12, 12, ... leading to the cylinder chamber SR. The remaining four oil chambers are connected with a left side cylinder chamber SL through oil introducing holes, and form second oil feed chambers 13, 13, ... leading to the cylinder chamber SL. Therefore, oil paths leading to the oil discharge chamber 11 through the first oil feed chamber 12 or second oil feed chamber 13 are formed on both sides of the oil supply chamber 10. Further, the oil supply chamber 10 is in communication with the oil feed chambers 12 and 13 through the throttle portion 6a, and the oil discharge chamber 11 is in communication with the oil feed chambers 12 and 13 through the throttle portion 6b.

[0035]

A hydraulic control valve constructed as above is a basic construction of an existing hydraulic control valve. However, in the present invention, periphery portions of the second oil grooves 5 forming the throttle portions 6a, 6a between the oil supply chamber 10 and the first and second oil feed chambers 12, 13 are respectively provided with chamfer portions 7, 7 for adjusting the throttle areas, and periphery portions of the second oil grooves 5 forming the throttle portions 6b, 6b between the oil discharge chamber 11 and the oil feed chambers 12, 13 are not provided with chamfer portions for adjusting the throttle areas. Therefore, the flow rate is controlled by the four

throttle portions 6a, 6a, ... provided with the chamfer portions 7.

[0036]

Each of the chamfer portions 7 is formed such that it has a predetermined inclined angle with respect to the peripheral surface of the valve spool 2 and with this design, the chamfer portion 7 has a predetermined width in the circumferential direction.

[0037]

The hydraulic control valve of the present invention is suitably used in a power steering apparatus in which the number of revolution of an electric motor M for driving the hydraulic pump P and the flow rate of the hydraulic pump P based on the steering angular velocity are increased or reduced, the flow rate to be controlled of the hydraulic control valve V can be abruptly changed from the low or zero flow rate which is smaller than the conventional minimum flow rate to be controlled to the high flow rate.

[0038]

FIGS. 2 are explanatory views of operation of the oil feed chamber, the oil supply chamber, and the oil discharge chamber shown in straightly developed manner, which are arranged along a fitted peripheral surfaces of the valve body and the valve spool in the hydraulic control valve. FIG. 2(a) shows a state (neutral state) in which the relative angular displacement is not generated between the valve body and the valve spool.

[0039]

In the state of standby control in which the steering wheel is not operated such as idling state, an electric motor M is stopped, or driven at extremely low speed. Therefore, the flow rate of oil

introduced from the hydraulic pump P to the oil supply chamber 10 becomes small less than 1 liter/min, and this 1 liter/min is equally divided into four, and the flow rate of about 0.25 liter/min is distributed to one oil supply chamber 10. The pressurized oil distributed to the four oil supply chambers 10 is equally divided into both side oil paths of the four oil supply chambers 10, and reaches the oil discharge chambers 11, 11 through the first oil feed chamber 12 or the second oil feed chamber 13, and flows into the hollow portion inside of the valve spool 2 through the oil discharge holes which are opened at the oil discharge chambers 11, 11, and merge in the hollow portion and circulates into the oil tank T. That is, pressure difference is not generated between the oil feed chambers 12 and 13 and between the cylinders SR and SL of the hydraulic cylinder S which is connected with the first oil feed chamber 12 and the second oil feed chamber 13 respectively, and the hydraulic cylinder S does not generate any force.

[0040]

FIG. 2(b) shows a state in which the steering wheel is operated and the relative angular displacement is generated between the valve body and the valve spool.

On the other hand, when the relative angular displacement is generated between the valve body 1 and the valve spool 2 as the steering wheel (not shown) is operated, throttle areas of the throttle portions 6a, 6a, ... between the oil supply chamber 10 and the first and second oil chambers 12, 13, and throttle areas of the throttle portions 6b, 6b, ... between the oil discharge chamber 11 and the first and second oil chambers 12, 13 are changed.

[0041]

This change is generated in opposite directions from each other at both sides of the oil chambers 12, 13. For example, the relative rotation of the valve spool 2 with respect to the valve body 1 is generated in the clockwise direction in FIG. 1, the valve spool 2 relatively moves in the direction shown with the hollow arrow in FIG. 2(b). At that time, on the both sides of the first oil feed chamber 12, the throttle area of the throttle portion 6a of the oil supply chamber 10 side is increased, and the throttle area of the throttle portion 6b of the oil discharge chamber 11 side is reduced without being controlled by the chamfer portion. On the other hand, on the both sides of the second oil feed chamber 13, the throttle area of the throttle portion 6a of the oil supply chamber 10 side is controlled and reduced by the chamfer portion 7, and the throttle area of the throttle portion 6b of the oil discharge chamber 11 side is increased.

[0042]

Therefore, in the beginning of the steering operation, the flow rate of about 0.25 liter/min is controlled by the chamfer portion 7 facing the throttle portions 6a between the oil supply chamber 10 and the second oil feed chamber 13. On the other hand, the flow rate is not controlled in the throttle portion 6b between the oil discharge chamber 11 and the first oil feed chamber 12 like the conventional example. That is, in the structure in which the eight first oil grooves 4 and the second oil grooves 5 are disposed at equal distances from one another, it is possible to reduce the number of portions where the flow rate is controlled can be reduced to half, i.e., four. Therefore, even through the eight oil grooves are disposed, the number of portions where the flow rate of pressurized oil introduced into the oil supply

chamber 10 is controlled is four, and pressurized oil is not distributed to the remaining four portions like the conventional example. Therefore, the amount of distribution per one portion where the flow rate is controlled is doubled as compared with the conventional example having eight oil grooves disposed at equal distances from one another. In the conventional example having eight oil grooves disposed at equal distances from one another, the flow rate in one portion where the flow rate is controlled becomes excessively small, and it is not possible to stabilize the hydraulic characteristics. However, in the present invention, since the flow rate doubled as compared with the conventional example can be controlled in one portion where the flow rate is controlled, it is possible to stabilize the hydraulic characteristics with the small or zero flow rate, and it is possible to eliminate discontinuity of the hydraulic characteristics.

[0043]

Further, the flow rate is controlled by the chamfer portion 7 facing the throttle portion 6a between the oil supply chamber 10 and the second oil feed chamber 13 as described above, and the flow rate is not controlled by the throttle portion 6b between the oil discharge chamber 11 and the first oil feed chamber 12 like the conventional example. Therefore, almost all the pressurized oil introduced into the oil supply chamber 10 as the steering angular velocity is increased is introduced into the first oil feed chamber 12 through the throttle portion 6a whose throttle area is increased, and the oil is supplied to the right side cylinder chamber SR which is in communication with the first oil feed chamber 12 and therefore, the oil pressure is abruptly increased as shown in FIG. 11.

[0044]

As the pressurized oil flows as described above, while the internal pressure of the first oil feed chamber 12 is approximately equivalent to that of the oil supply chamber 10, an internal pressure of the second oil feed chamber 13 is reduced by an amount corresponding to pressure reduction caused by communication with the throttle portion 6a whose throttle area is reduced between the second oil feed chamber 13 and the oil supply chamber 10, a pressure difference is generated between the first and second oil chambers 12, 13 and between the cylinder chambers SR and SL which are in communication with the first and second oil chambers 12, 13, and the hydraulic cylinder S generates the hydraulic force (steering assist force) flowing from the cylinder chamber SR to the cylinder chamber SL.

[0045]

With such an operation of the hydraulic cylinder S, operating oil in the cylinder chamber SL is pushed out from the cylinder chamber SL, and circulated into the second oil feed chamber 13 which is connected with the cylinder chamber SL, merges with the operating oil from the oil supply chamber 10, and is introduced into the oil discharge chamber 11 through the throttle portion 6b whose throttle area is increased in the other side of the second oil feed chamber 13, and is discharged into the oil tank T through the hollow portion of the valve spool 2.

[0046]

The steering assist force generated by the hydraulic cylinder S by the above-described operation depends on a degree of reduction in throttle areas of the throttle portion 6a between the oil supply chamber

10 and the second oil feed chamber 13, and the throttle portion 6b between the oil discharge chamber 11 and the first oil feed chamber 12. Here, the degree of reduction in throttle areas of the throttle portions 6a and 6b correspond to the relative angular displacement between the valve body 1 and the valve spool 2, and this relative angular displacement corresponds to magnitude of the steering angle. Therefore, the hydraulic cylinder S generates the steering assist force corresponding to the magnitude of the steering angle.

[0047]

In the operational state shown in FIG. 2(b), pressurized oil flow at high speed through the throttle portion 6a whose throttle area is reduced between the oil supply chamber 10 and the second oil feed chamber 13, by the influence of the pressure difference at opposite sides of the throttle portion 6a. There, the chamfer portions 7, 7 are disposed at the pair of throttle portions 6a, 6a of the four second oil grooves 5 forming the oil supply chamber 10 and with this design, the pressurized oil in the oil supply chamber 10 flows from the second oil groove 5 toward the first oil groove 4 along the chamfer portions 7,7. Therefore, it is possible to gradually and straightly reduce the hydraulic force of the operating oil flowing through the throttle portions 6a, 6a along the chamfer portions 7, 7, and to prevent the generation of cavitation.

[0048]

Further, when the relative rotation of the valve spool 2 with respect to the valve body 1 is generated in the counterclockwise direction in FIG. 1 which is opposite from the above-described case, the throttle area of the throttle portion 6a of the oil supply chamber 10 is

increased on the both sides of the second oil feed chamber 13, and the throttle area of the throttle portion 6b of the oil discharge chamber 11 is reduced without being controlled by the chamfer portion. Whereas, on the both sides of the first oil feed chamber 12, the throttle area of the throttle portion 6a of the oil supply chamber 10 is controlled and reduced by the chamfer 7, and the throttle area of the throttle portion 6b of the oil discharge chamber 11 is increased. Therefore, almost all the pressurized oil introduced into the oil supply chamber 10 is introduced mainly to the second oil feed chamber 13 through the throttle portion 6a whose throttle area is increased, and is supplied to the cylinder chamber SL which is in communication with the second oil feed chamber 13, and the hydraulic cylinder S generates the hydraulic force (steering assist force) flowing from the cylinder chamber SL to the cylinder chamber SR.

[0049]

Second Embodiment

FIG. 3 is a schematic transverse sectional view showing a hydraulic control valve according to the present invention, together with a hydraulic circuit of a power steering apparatus.

The hydraulic control valve of the second embodiment is constructed so that second oil grooves 5, 5 of the second valve spool 2 form the first and second oil feed chambers 12 and 13 whereas the first oil grooves 4, 4 of the valve body 1 alternately form the oil supply chamber 10 and the oil discharge chamber 11. In this construction, the edge portions of the second oil grooves 5, 5 forming the throttle portions 6b, 6b between the oil discharge chamber 11 and the first and second oil feed chambers 12, 13 are provided with the chamfer portions

7, 7 for adjusting the throttle area, and edge portions of the second oil grooves 5, 5 forming the throttle portions 6a, 6a between the oil supply chamber 10 and the first and second oil chambers 12, 13 are not provided with the chamfer portions for adjusting the throttle area. Therefore, the flow rate is controlled by the four throttle portions 6b, 6b, ... provided with the chamfer portions 7, 7. Other structure and operation are the same as those of the first embodiment shown in FIGS. 1 and 2, elements similar to those of the first embodiment are represented by the same reference numerals, and explanation of its detailed structure and operation will be omitted.

[0050]

FIGS. 4(a) and (b) are explanatory views of operation of an oil feed chamber, an oil supply chamber, and an oil discharge chamber shown in straightly developed manner, which are arranged along a fitted peripheral surfaces of a valve body and a valve spool. FIG. 4(a) shows a state (neutral state) in which the relative angular displacement is not generated between the valve body and the valve spool. FIG. 4(b) shows a state in which the steering wheel is operated and the relative angular displacement is generated between the valve body and the valve spool.

[0051]

In the second embodiment, for example, when the relative rotation of the valve spool 2 with respect to the valve body 1 is generated in the clockwise direction in FIG. 3, the valve spool 2 relatively moves in the direction shown with the hollow arrow in FIG. 4(b). At that time, on the both sides of the second oil feed chamber 13, the throttle area of the throttle portion 6a of the oil supply chamber 10

is increased, and the throttle area of the throttle portion 6b of the oil discharge chamber 11 is controlled and reduced by the chamfer portion. On the other hand, on the both sides of the first oil feed chamber 12, the throttle area of the throttle portion 6a of the oil supply chamber 10 is reduced without being controlled by the chamfer portion, and the throttle area of the throttle portion 6b of the oil discharge chamber 11 is increased.

[0052]

Therefore, in the beginning of the steering operation, the flow rate is controlled by the chamfer portion 7 facing the throttle portions 6b between the oil discharge chamber 11 and the second oil feed chamber 13. On the other hand, the flow rate is not controlled in the throttle portion 6a between the oil supply chamber 10 and the first oil feed chamber 12 like the conventional example. That is, even through the eight oil grooves 4, 5 are disposed at equal distances, the number of portions where the flow rate is controlled can reduced to four which is the half, similarly to the first embodiment.

[0053]

Here, the chamfer portions 7, 7 provided at the throttle portions 6b, 6b are disposed such as to face the pair of throttle portions 6b, 6b of the four first oil grooves 4 forming the oil discharge chamber 11 and with this design, the pressurized oil of the oil supply chamber 10 and the second oil feed chamber 13 flows from the second oil grooves 5 toward the first oil grooves 4 along the chamfer portions 7. Therefore, it is possible to gradually and straightly reduce the pressure of the operating oil flowing through the throttle portions along the chamfer portions 7, 7, and to reduce the generation of cavitation as shown in

the first embodiment.

[0054]

(Third Embodiment)

FIGS. 5(a) and 5(b) are explanatory views of operation of an oil feed chamber, an oil supply chamber, and an oil discharge chamber shown in straightly developed manner, which are arranged along a fitted peripheral surfaces of a valve body and a valve spool. FIG. 5(a) shows a state (neutral state) in which the relative angular displacement is not generated between the valve body and the valve spool. FIG. 5(b) shows a state in which the steering wheel is operated and the relative angular displacement is generated between the valve body and the valve spool.

[0055]

The hydraulic control valve of the third embodiment is constructed so that the first oil grooves 4,4 of the first valve body 1 form the first and second oil feed chambers 12 and 13 whereas the second oil grooves 5, 5 of the valve spool 2 alternately form the oil supply chamber 10 and the oil discharge chamber 11. In this construction, the edge portions of the first oil grooves 4 forming the throttle portions 6a, 6a between the oil supply chamber 10 and the first and second oil feed chambers 12, 13 are provided with the chamfer portions 7, 7 for adjusting the throttle areas, whereas the edge portions of the first oil grooves 4, 4 forming the throttle portions 6b, 6b between the oil discharge chamber 11 and the first and second oil chambers 12, 13 are not provided with the chamfer portions for adjusting the throttle areas. Therefore, the flow rate is controlled by the four throttle portions 6a, 6a, ... provided with the chamfer portions 7, 7.

Other structure and operation are the same as those of the first embodiment shown in FIGS. 1 and 2, elements similar to those of the first embodiment are represented by the same reference numerals, and explanation of its detailed structure, operation and effects will be omitted.

[0056]

(Fourth Embodiment)

FIGS. 6(a) and 6(b) are explanatory views of operation of an oil feed chamber, an oil supply chamber, and an oil discharge chamber shown in straightly developed manner, which are arranged along a fitted peripheral surfaces of a valve body and a valve spool. FIG. 6(a) shows a state (neutral state) in which the relative angular displacement is not generated between the valve body and the valve spool. FIG. 6(b) shows a state in which the steering wheel is operated and the relative angular displacement is generated between the valve body and the valve spool.

[0057]

The hydraulic control valve of the fourth embodiment is constructed so that second oil grooves 5, 5 of the second valve spool 2 form the first and second oil feed chambers 12 and 13 whereas the first oil grooves 4, 4 of the valve body 1 alternately form the oil supply chamber 10 and the oil discharge chamber 11. In this construction, the edge portions of the second oil grooves 5, 5 forming the throttle portions 6b, 6b between the oil discharge chamber 11 and the first and second oil feed chambers 12, 13 are provided with the chamfer portions 7, 7 for adjusting the throttle area, and edge portions of the second oil grooves 5, 5 forming the throttle portions 6a, 6a between the oil supply

chamber 10 and the first and second oil chambers 12, 13 are not provided with the chamfer portions for adjusting the throttle area. Therefore, the flow rate is controlled by the four throttle portions 6b, 6b, ... provided with the chamfer portions 7, 7. Other structure and operation are the same as those of the first embodiment shown in FIGS. 1 through 4, elements similar to those of the first embodiment are represented by the same reference numerals, and explanation of its detailed structure, operation and effects will be omitted.

[0058]

Although the hydraulic control valve for controlling the oil pressure to be supplied to the hydraulic cylinder S of the power steering apparatus has been described above in each of the embodiments, the present invention is not limited to this, and it is of course possible to apply the present invention to all rotary-type hydraulic control valve having a valve body 1 and a valve spool 2 fitted to each other, and provided around their fitting peripheral surfaces with a plurality of throttle portions whose throttle areas are changed in accordance with relative angular displacement between the valve body and the valve spool.

[0059]

[Advantages of the Invention]

As described above in detail, according to the first aspect of the present invention, the chamfer portions are formed only on the throttle portions between the oil supply chambers and the oil feed chambers or on the throttle portions between the oil discharge chambers and the oil feed chambers, and the number of portion where the flow rate is controlled by the chamfer portions is reduced to half of

the existing structure. Therefore, even if the six or more oil grooves are disposed at equal distances from one another, the quantity of oil distributed to one portion where the flow rate is controlled when the minimum flow rate to be controlled is reduced as small as possible is doubled as compared with the conventional structure. Therefore, it is possible to stabilize the hydraulic characteristics in a region where the flow rate to be controlled is minimum, and to eliminate the discontinuity of the hydraulic characteristics. Accordingly, the flow rate of the hydraulic pump is increased on the basis of the steering angular velocity of the steering wheel so as to increase the flow rate of the hydraulic control valve from a flow rate as smaller as possible compared with the conventional minimum flow rate or zero flow rate to a high flow rate. With this structure, the hydraulic characteristics are stable, and there is no discontinuity of the hydraulic characteristics in a region where the flow rate to be controlled is minimum, and it is possible to inexpensively produce a hydraulic control valve which eliminates discontinuity in the steering torque, by using an existing hydraulic control valve having six or more oil grooves disposed at equal distances from one another.

[0060]

According to the second aspect of the present invention, it is possible to flow the operating oil introduced in the oil supply chamber while gradually and straightly reducing the pressure from the second oil groove toward the first oil groove along the chamfer, and it is possible to reduce flowing noise which is generated when the relative angular displacement between the valve body and the valve spool is great.

[0061]

According to the third aspect of the present invention, in the case where the hydraulic control valve is based on the first aspect of the invention, the flow rate of the hydraulic pump is increased on the basis of the steering angular velocity of the steering wheel so as to abruptly increase the flow rate of the hydraulic control valve from a flow rate as smaller as possible than the conventional minimum flow rate or zero flow rate to a high flow rate. Further, with this structure, there is no discontinuity of the hydraulic characteristics in a region where the flow rate to be controlled is minimum, and it is possible to inexpensively produce a hydraulic control valve which eliminates discontinuity in the steering torque, by using an existing hydraulic control valve having six or more oil grooves disposed at equal distances from one another. Moreover, in the case where the hydraulic control valve is based on the second aspect of the invention, it is possible to reduce flowing noise which is generated when the relative angular displacement between the valve body and the valve spool is great.

[0062]

According to the fourth aspect of the present invention, the flow rate of the hydraulic pump is increased on the basis of the steering angular velocity of the steering wheel so as to abruptly increase the flow rate of the hydraulic control valve from a flow rate as smaller as possible than the conventional minimum flow rate to a high flow rate. With this structure, there is no discontinuity of the hydraulic characteristics in a region where the flow rate to be controlled is minimum, and it is possible to inexpensively produce a hydraulic control valve which eliminates discontinuity in the steering torque, by

using an existing hydraulic control valve having six or more oil grooves disposed at equal distances from one another. Moreover, it is possible to stop the operation of the electric motor or to drive the motor at extremely low speed. Therefore, it is possible reduce the energy consumption while the steering operation is not conducted.

[Brief Description of the Drawings]

[FIG. 1] A schematic transverse sectional view showing a first embodiment of a hydraulic control valve according to the present invention

[FIGS. 2] Explanatory views of operation of a hydraulic control valve of the present invention

[FIG. 3] A schematic transverse sectional view showing a second embodiment of a hydraulic control valve according to the present invention

[FIGS. 4] Explanatory views of operation of a hydraulic control valve according to the present invention

[FIGS. 5] Explanatory views of operation of a hydraulic control valve according to the third aspect of the present invention

[FIGS. 6] Explanatory views of operation of a hydraulic control valve according to the fourth aspect of the present invention

[FIG. 7] A schematic transverse sectional view showing a prior art hydraulic control valve

[FIG. 8] Explanatory views of operation of the prior art hydraulic control valve

[FIG. 9] A schematic transverse sectional view showing a prior art hydraulic control valve

[FIG. 10] A view of flow rate characteristics showing a relation

between steering angular velocity of a steering wheel and flow rate of a pump

[FIG. 11] A view of hydraulic characteristics showing a relation between steering torque applied to a steering wheel and hydraulic force controlled by the hydraulic control valve

[FIG. 12] A view for explaining a state where the hydraulic characteristics of the hydraulic control valve become discontinuous

[Description of the Reference Numerals]

1	Valve Body
2	Valve Spool
4	First Oil Groove
5	Second Oil Groove
6a, 6b	Throttle Portion
7	Chamfer Portion
10	Oil Supply Chamber
11	Oil Discharge Chamber
12	First Oil Feed Chamber
13	Second Oil Feed Chamber
P	Hydraulic Pump
S	Hydraulic Cylinder

[Name of the Document] Abstract of the Disclosure

[Abstract]

[Purpose] With an existing hydraulic control valve having six or more oil grooves disposed at equal distances from one another, it is possible to stabilize the hydraulic characteristics in a region where the flow rate to be controlled is minimum, and to eliminate the discontinuity of the hydraulic characteristics in the region.

[Means for Accomplishing the Purpose] The chamfer portions are provided only on throttle portions 6a between oil supply chambers 10 and oil feed chambers 12 and between the oil supply chambers 10 and oil feed chambers 13, or only throttle portions 6b between oil discharge chambers 11 and the oil feed chambers 12 and between the oil discharge chambers 11 and the oil feed chambers 13. With this design, even if six or more oil grooves are disposed at equal distances from one another, when the minimum flow rate to be controlled is reduced as small as possible, a distribution amount for one portion where the flow rate is controlled can be doubled as compared with an existing structure, and it is possible to eliminate the discontinuity of the hydraulic characteristics in a region where the flow rate to be controlled is minimum.

[Drawing to Be Selected] FIG. 1

FIG. 1

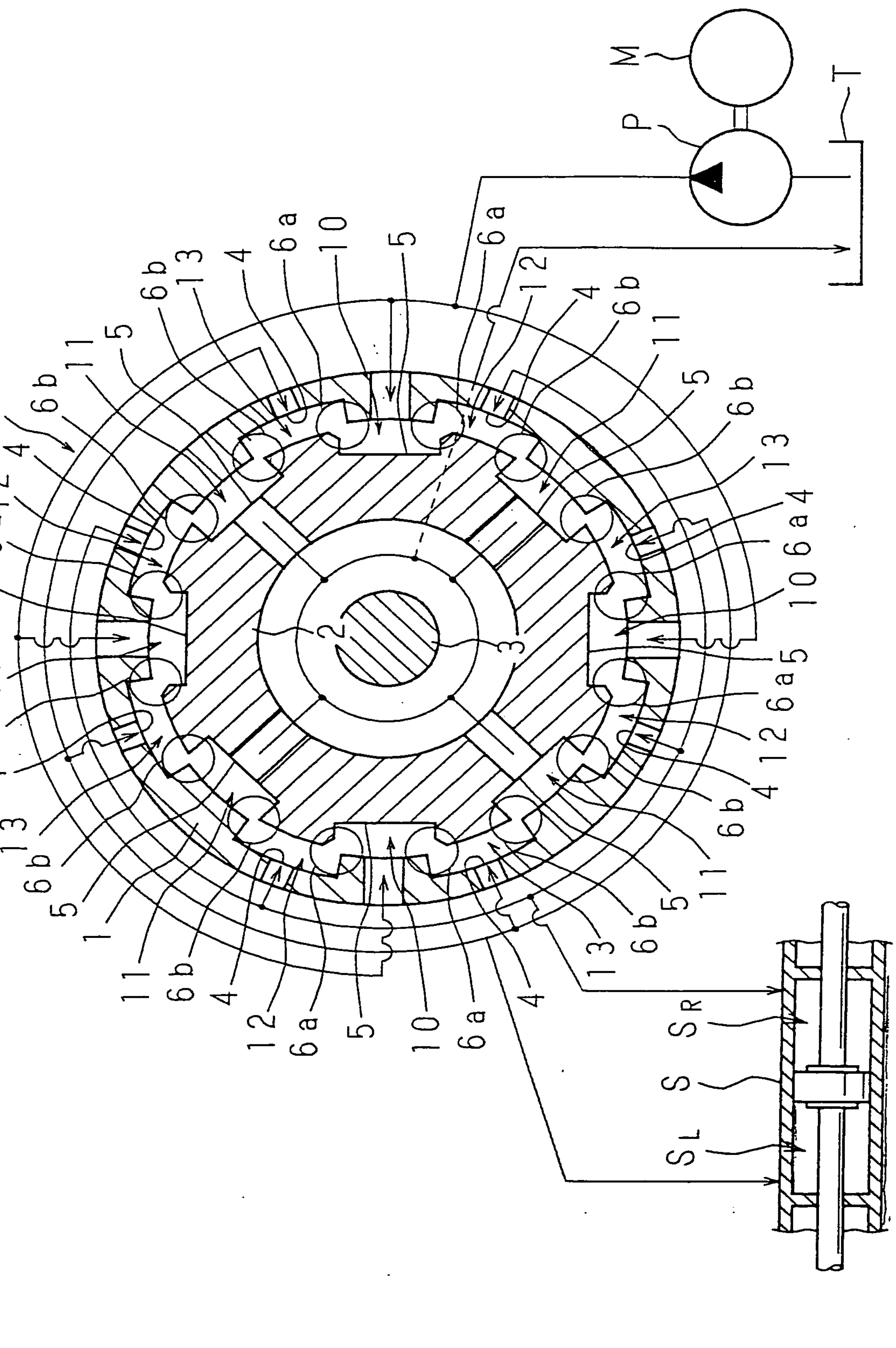
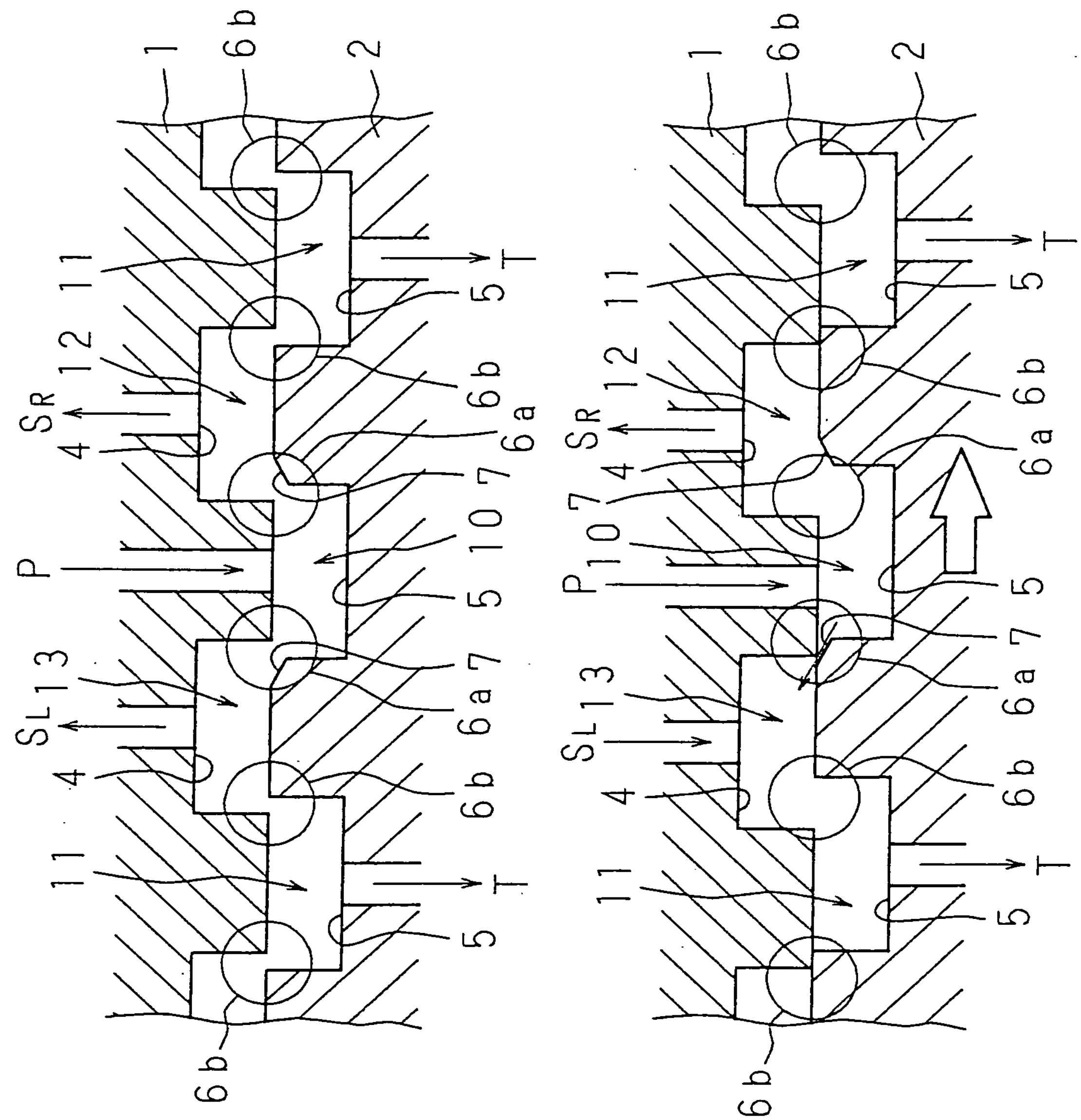


FIG. 2



(a)

(b)

FIG. 3

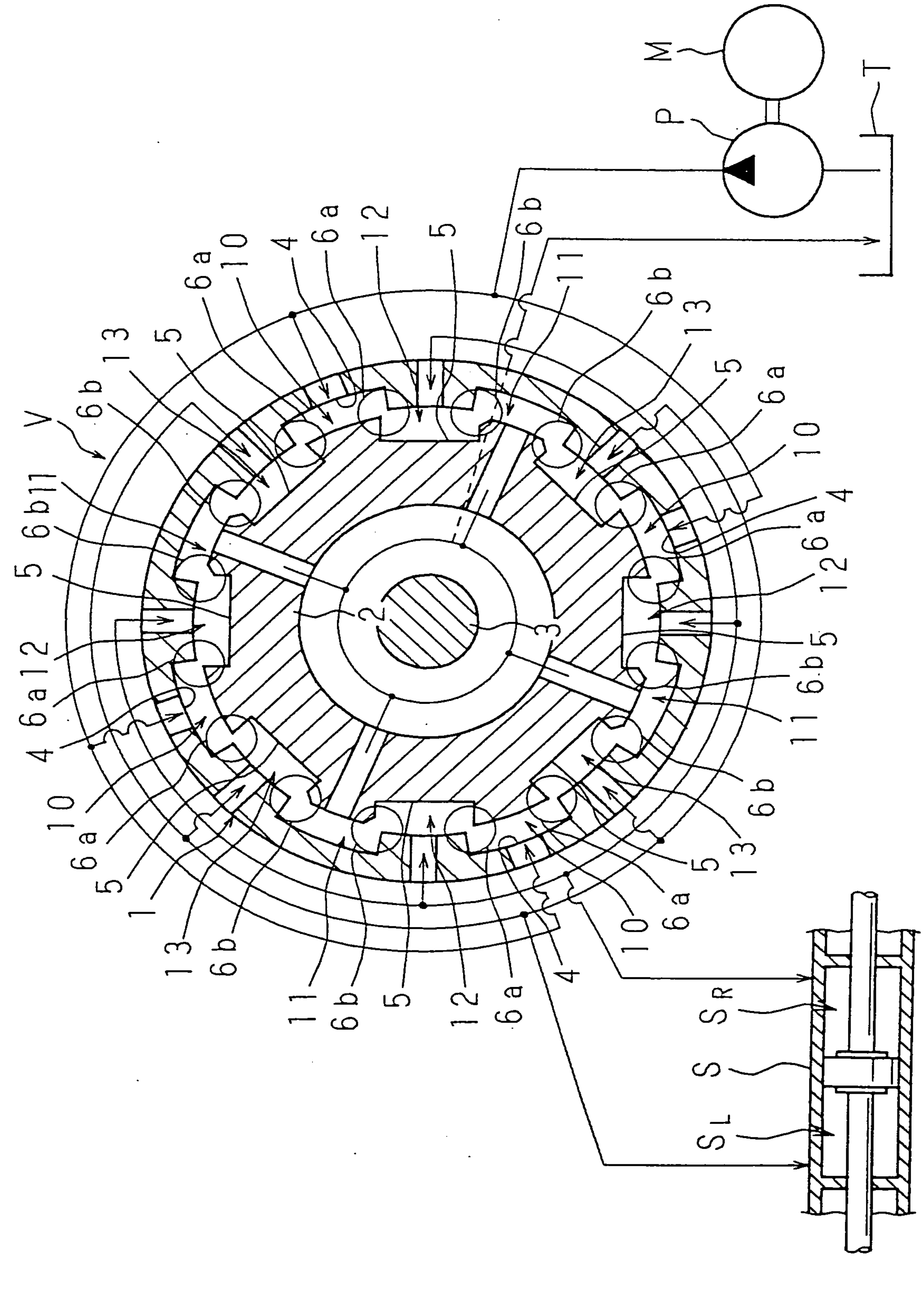
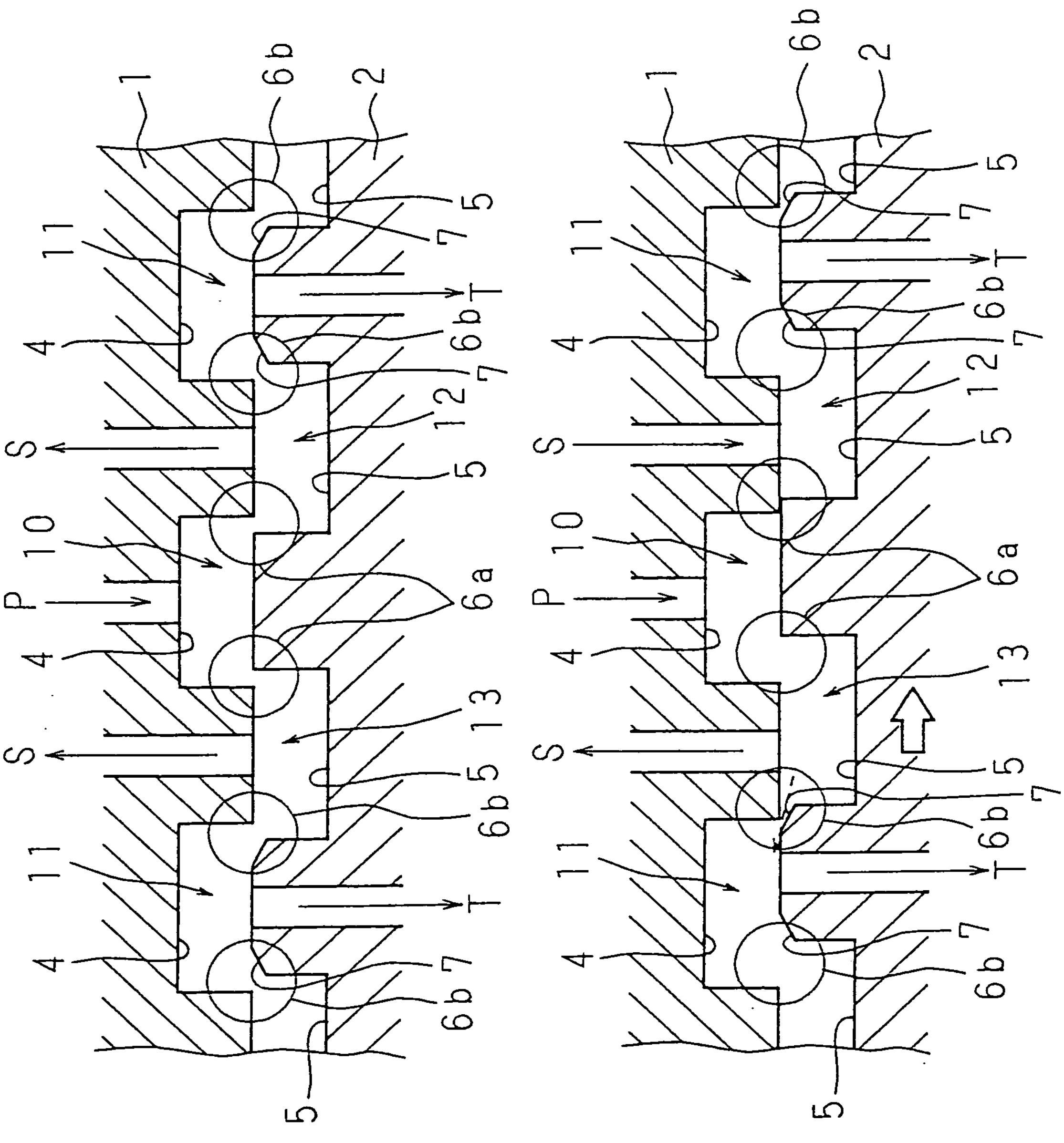


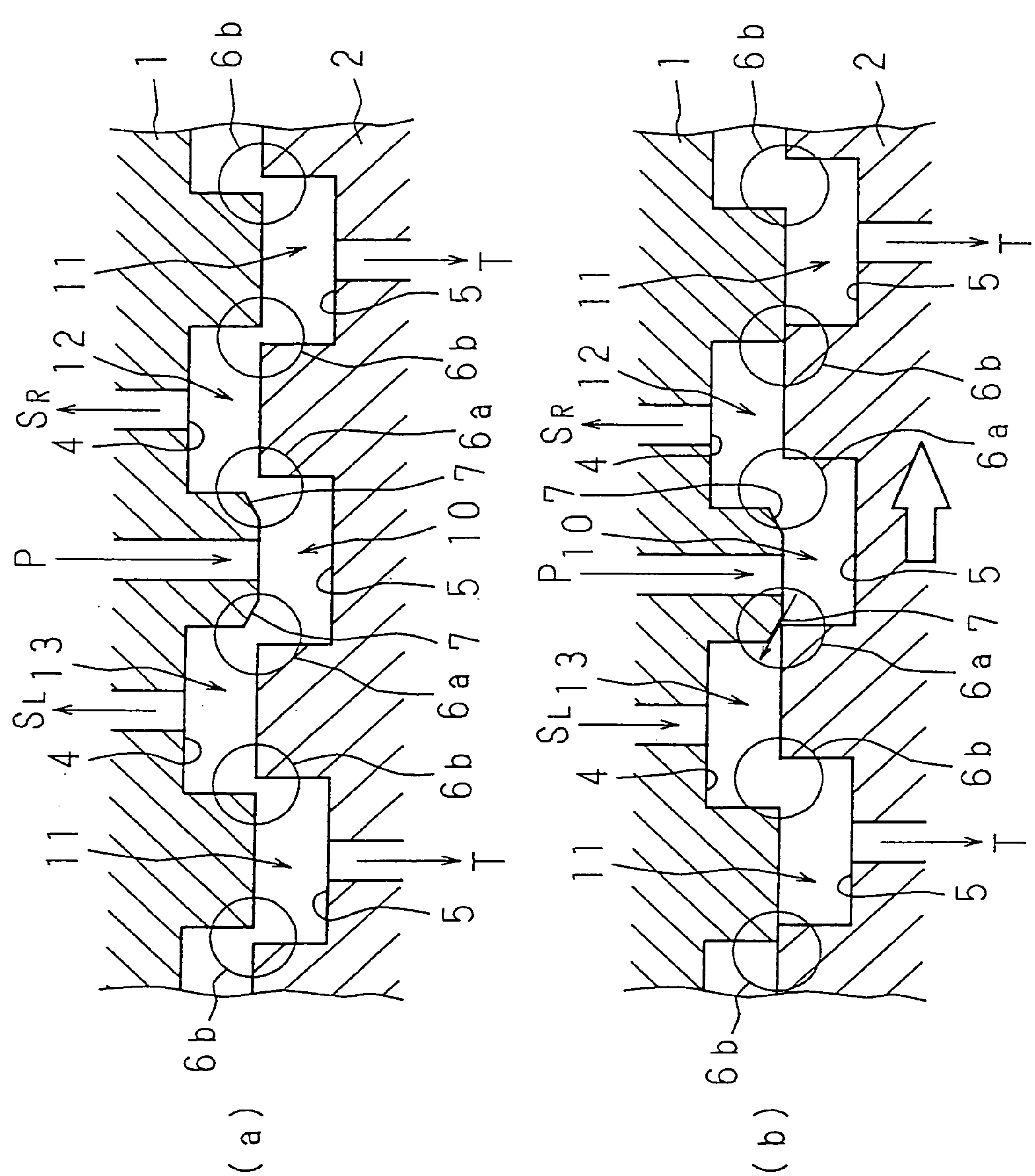
FIG. 4



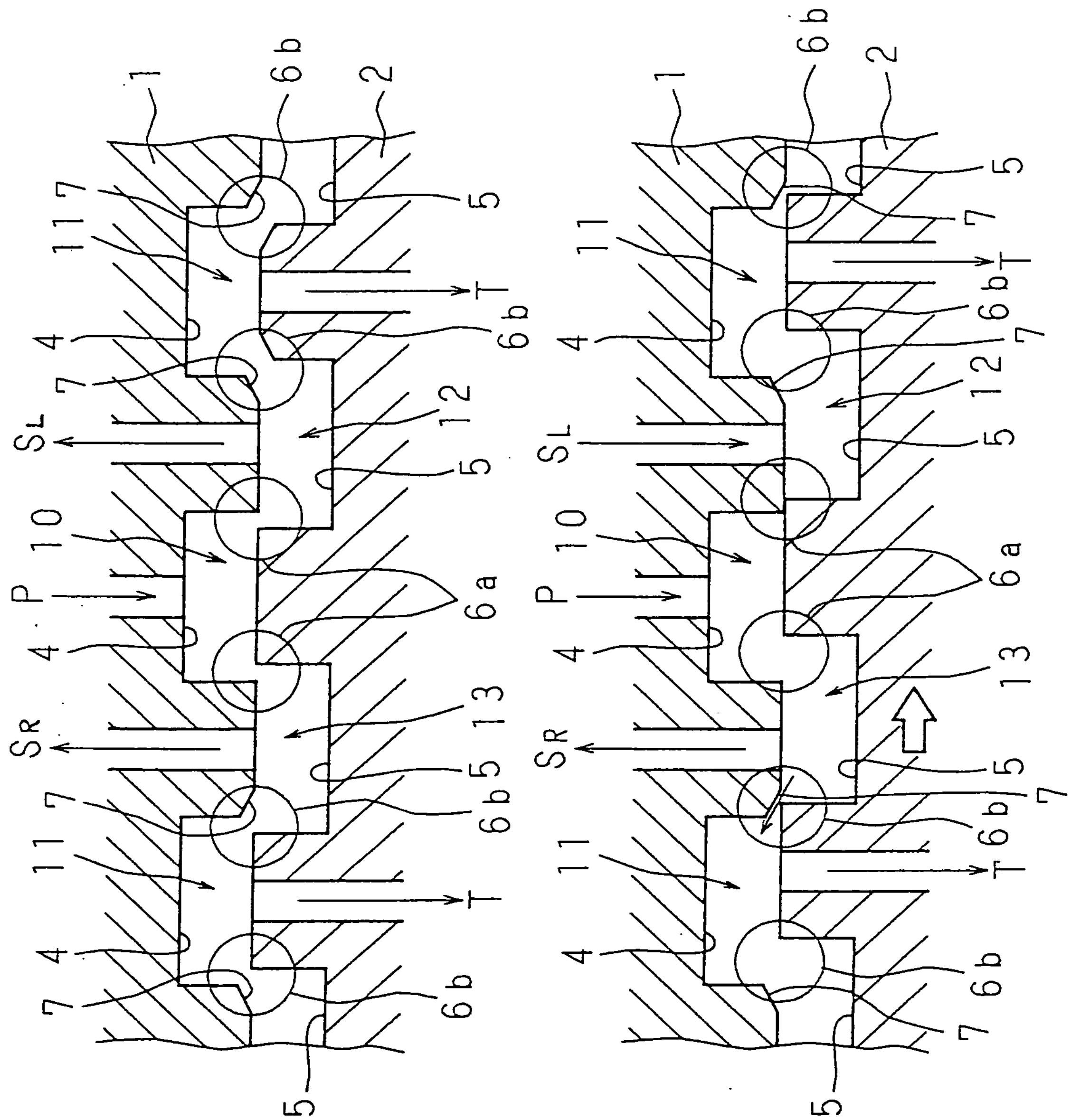
(a)

(b)

FIG. 5



F I G. 6



(a)

(b)

FIG. 7

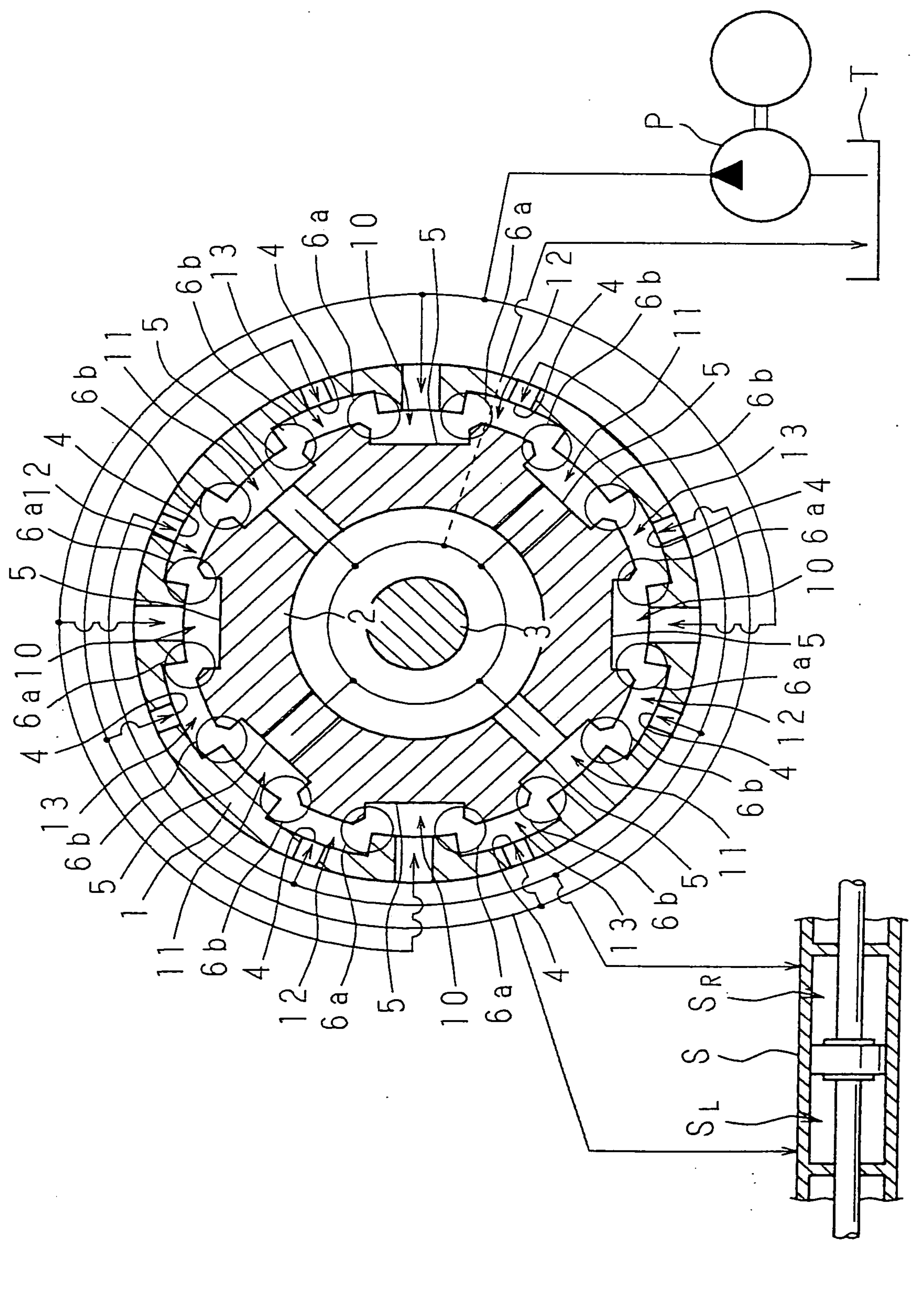


FIG. 8

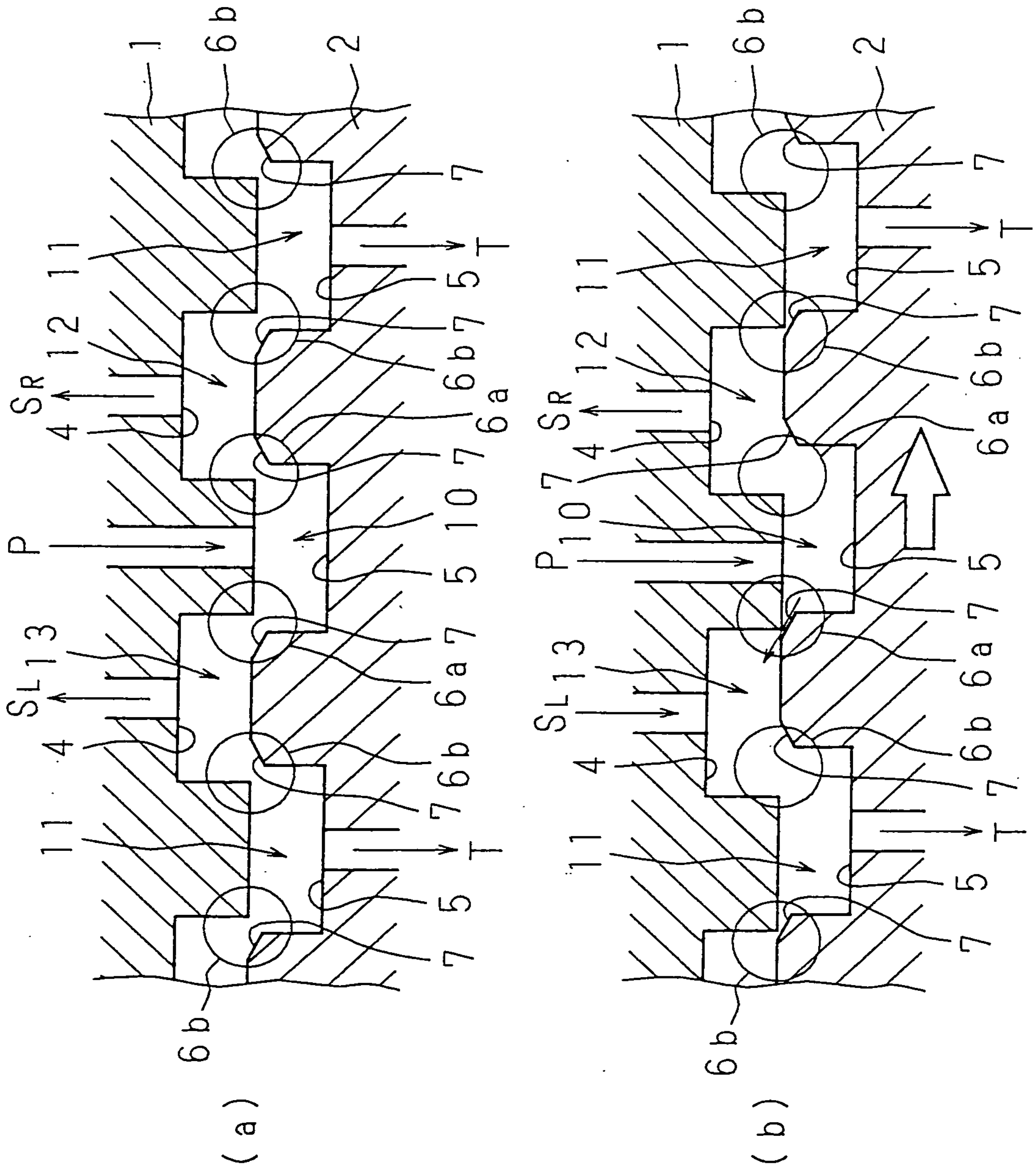


FIG. 9

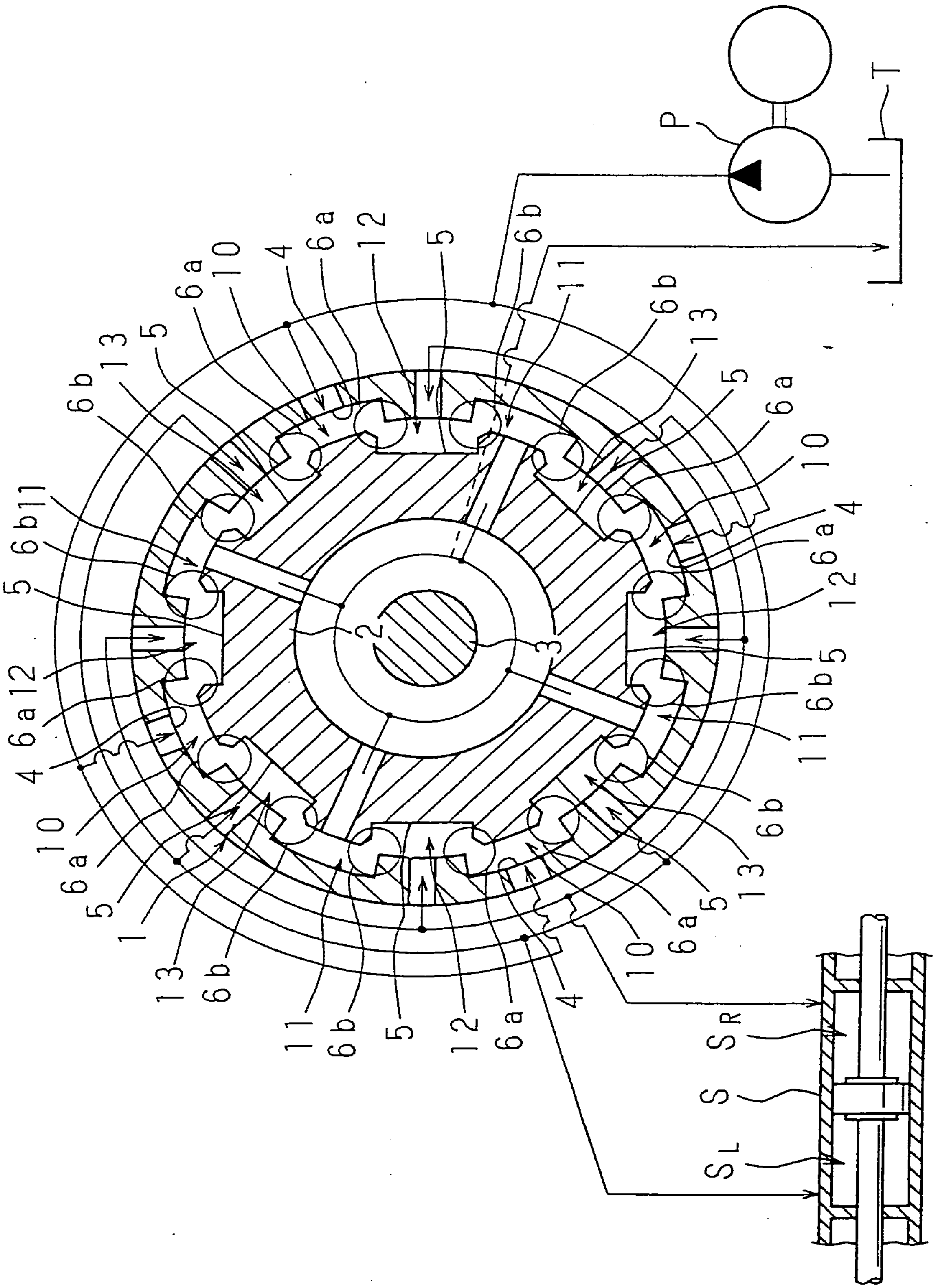


FIG. 10

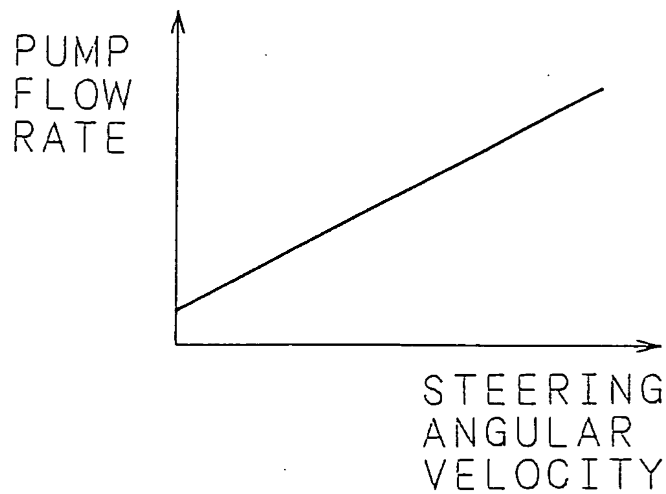


FIG. 11

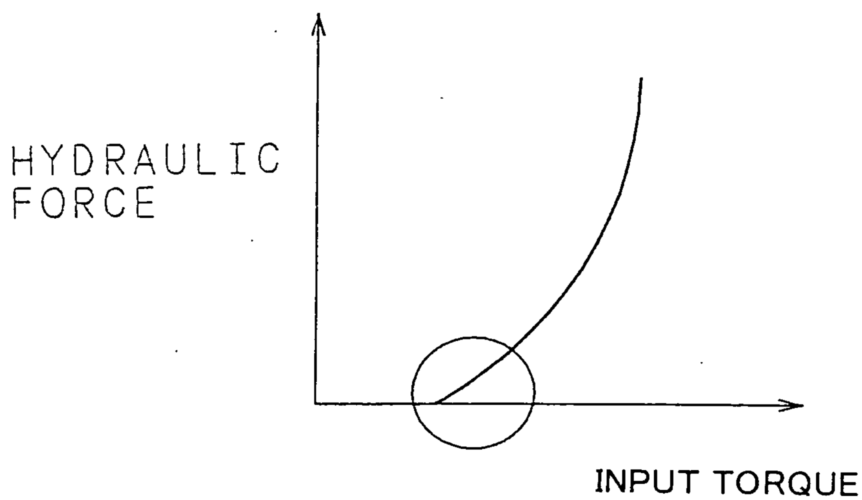


FIG. 12